465

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PROCEEDINGS

AMERICAN SOCIETY OF CIVIL ENGINEERS

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DISCUSSION OF PROCEEDINGS - SEPARATES

273, 294

POWER DIVISION

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DISCUSSION OF CHARTS FOR DESIGNING AIR CHAMBERS FOR PUMP DISCHARGE LINES PROCEEDINGS-SEPARATE NO. 273

C. E. WITHERS[1].—Mr. Evans and Mr. Crawford have done an excellent job of presenting their "Charts for Designing Air Chambers for Pump Discharge Lines." A check of a number of points on the charts by using Mr. Angus's graphical solution for air chambers will show that the charts are sufficiently accurate, that for all practical purposes, the graphical solution need not be resorted to unless the designer wishes to use it for checking or to obtain some information about the transient hydraulic conditions in the system that are not indicated by the charts. If the authors of this paper could produce the results of any tests that have been conducted on a pump system with an air chamber that would not only substantiate the accuracy of their charts but would also prove that the entire conception of the transient hydraulic conditions in a pump system of this type are also correct, they would contribute materially to the value of their charts.

It has been pointed out in this paper that for successful operation of an air chamber in a pump discharge line certain minimum requirements for the capacity of air and water in the tank must be maintained. This indicates the importance of being able to maintain the water level in the air chamber under all conditions of operation and the necessity of automatic shutting down the pumps if the proper water level cannot be maintained in the tank. Inasmuch as successful air chamber operation is so dependent on successful water level control, this item deserves more attention.

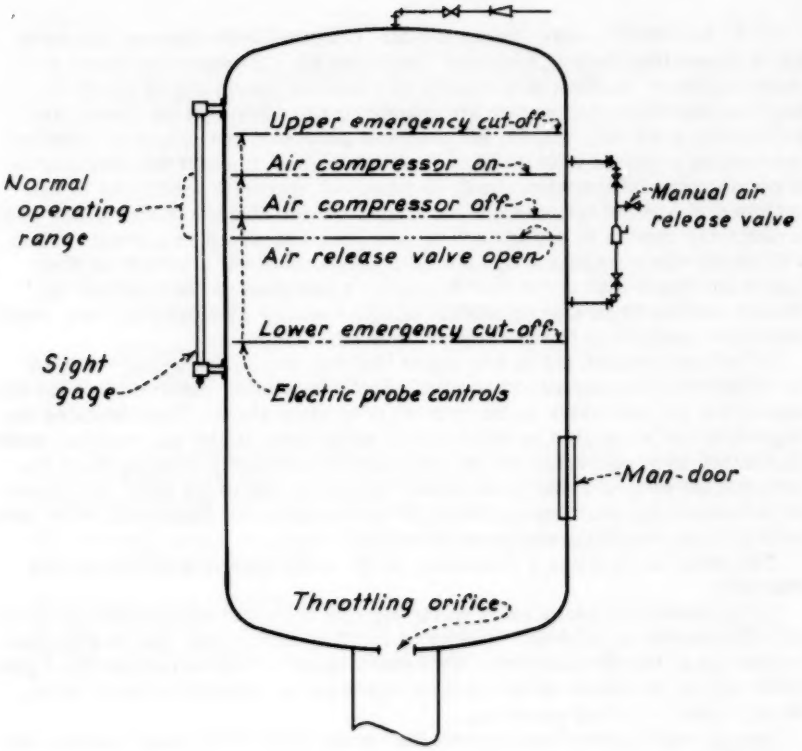
The attached figure is a schematic of the water level control for an air chamber.

First, there is a check valve in the air line from the compressor to eliminate the possibility of water backing up into the compressor due to any malfunctioning of the air chamber. This check valve should be guarded by a gate valve so that the check valve could be inspected or repaired without taking the air chamber out of operation.

Second, sight gages should cover the entire water level range between the "upper emergency" water level to the "lower emergency" water level so that in the event of a pump shut-down due to the operation of the water level control system the operator can tell at a glance whether the shut-down was caused by high water level or low water level in the tank. These gages are also necessary when the tank is first placed into operation so that the operation of the water level control system can be checked and altered if necessary.

Third, comes the control of the water levels in the air chamber. These controls should be of the electric probe type in order to eliminate as many sources of malfunctioning as possible. The "upper emergency" water level switch should stop all the pumps and require manual restarting. This is necessary because when the air chamber fills with water, the protection to the system which it was designed to give has been eliminated. The cause of the malfunctioning must be removed before the pumps are started once more.

1. Mechanical Engineer, Denver, Colorado.



SCHEMATIC WATER LEVEL CONTROL
AIR CHAMBER DESIGN

Fig. 1

Next come the "compressor on" water level switch. Air leaks in the system and absorption of the air by the water will reduce the volume of air and it must be replaced by the compressor. This water level switch must be located far enough below the "upper emergency" point that when the water level is just below the "compressor on" point, any normal surge in the tank will not actuate the "upper emergency" water level switch. The "compressor off" water level switch should be located at some point just below the normal operating water level in the air chamber and far enough below the "compressor on" point that the compressor can operate for a reasonable time once it has been started. The "lower emergency" water level switch will shut-down all the pumps and should require manual restarting of the pumps. This "lower emergency" water level must be far enough below the "compressor off" water level that any normal surge in the tank will not actuate it. It must be high enough in the tank that when the water level gets down to this point and the pumps are shut-off the down surge will not drain all the water out of the tank allowing air to enter the pump discharge line. If after an air tank has been installed and it is determined that every power outage actuates the "lower emergency" water level switch, and this occurs with such frequency that the air chamber operation is objectionable, then a time delay switch can be installed so that the pumps can be started automatically, but this time delay must be short enough that the pump discharge lines will not have time to drain before the pumps are restarted.

Fourth. In some air chambers, particularly in installations for relatively low heads, it may be desirable to install a float operated air release valve just below the "compressor off" water level. This valve will release air from the chamber whenever the water level falls below this point and will prevent such factors as temperature change from resulting in an inadequate water level in the chamber. The size of this valve must be limited so that it will not release an excessive amount of air during a transient down surge following a power interruption and thereby render the air chamber ineffective.

Fifth. A manual air release valve should be installed between the "compressor off" and the "compressor on" water levels in order to provide some measure of manual control of the water in the tank during manual restarting of the pumps.

Any connection such as manholes should be located below the "lower emergency" water level if possible so that any leaks that develop can be readily detected.

When the transient hydraulic conditions can be controlled by an air chamber, a properly designed tank with the proper control of the water levels can be very effective.

HENRY M. PAYNTER,[2] A.M. ASCE.—The authors are to be commended for the excellent presentation of the results of their analysis in the form of charts capable of direct practical application by hydraulic engineers for the design of air chambers.

The writer has been interested in the air chamber problem as well as related surge and water hammer problems for some time (1) (2) (3) (4). He has determined to his satisfaction that the effects of the elasticity of the water column and pipe walls are negligible for the majority of pump discharge lines. This fact would imply that the pressures in the tank and along the pipe are

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predominantly functions of the inertial parameter σ^*, and that the elastic parameter ρ^* has only parasitic effects. This can be demonstrated by plotting the results as functions of σ^* instead of $2\rho^*\sigma^*$. The writer has replotted, in this manner, for example, Figure 3 of the paper as Figure D-1 of this discussion. Here it is evident that the quantity ρ^* has only slight effects until it becomes very small (corresponding to extremely high static lifts). Also shown on the plot are the analytical curves A, B, C, and D for ρ^* infinite (a rigid water column). The equation for these curves may be written:

$$\sigma^* = \frac{1}{(1+Z)^{1/6}[5+(1/1+Z)]-6} \qquad \text{------(D-1)}$$

where Z = (surge)/H_0^* and is defined as follows:

Curve A = Z Positive (upsurge)
Curve B = Z Negative (downsurge)
Curve C = (1/2) Z for Curve A
Curve D = (1/2) Z for Curve B

For the general case of K = 0, in terms of an arbitrary process exponent n, the relation between extreme surges and chamber size may be written as follows:

For $n \neq 1.0$ $$1/\sigma^* = \frac{1}{n-1}\left[(1+Z)^{\frac{n-1}{n}} - 1\right] + \left[\frac{1}{(1+Z)}\, 1/n\right] - 1$$

------(D-1)

For $n = 1.0$ $$1/\sigma^* = \ln(1+Z) + \left(\frac{1}{1+Z}\right) - 1$$

------(D-2)

These results were first published in two pioneering papers of another Italian engineering scientist, Professor Giuseppe Evangelisti of the University of Bologna (5) (6). Dr. Evangelisti also studied in exhaustive detail by ingenious graphical analysis, the effects of the process exponent n, the surge parameter σ^*, the pipe friction loss and the chamber throttling action.

The writer has obtained permission from Prof. Evangelisti to reproduce herewith, with English titles, certain of the many design charts resulting from his studies. The writer feels that these curves, together with those of the authors, will go far toward making the design of air chambers a more straightforward procedure.

It is necessary here to indicate the slightly different notation of Prof. Evangelisti as demonstrated in Figure D-2 of the discussion and in Table D-1 on the following page.

In summary, the Evans and Crawford parameter σ^* is based upon initial head and volume, while the Evangelisti parameter σ is merely the inverse of σ^* but based upon static values of head and volume.

The four Evangelisti charts presented in Figures D-3, D-4, D-5, and D-6, are adapted from the originals presented in Reference (6). They represent the upsurges and downsurges corresponding to the conditions given in Table D-2 on the following page.

The charts thus give the values of relative downsurge (at left) and upsurge (at right) corresponding to the values of the inertial parameter σ and the relative friction loss $h_o = H_o/Y_s$ where H_o is the initial friction loss and Y_s

Table D-1

COMPARATIVE TERMINOLOGY

No.	Term	NOTATION	
		Evans & Crawford	Evangelisti
(1)	Air volume	C	U
(2)	Initial velocity	V_o	V_o
(3)	Initial absolute pressure head	H_o^*	$Y_s + H_o$
(4)	Minimum head at pump for maximum downsurge	H_o^* - Downsurge	$Y_s - Z_{min}$
(5)	Maximum head at pump for maximum upsurge	H_o^* + Upsurge	$Y_s + Z_{max}$
(6)	Initial pipe friction loss	-----	H_o
(7)	Initial chamber throttling loss	-----	K_o
(8)	Total head loss	KQ_o^2	$H_o + K_o$
(9)	Chamber inertial parameter	$\sigma^* = \frac{2gHo^*Cg}{V_o^2AL}$	$\sigma = \frac{V_o^2AL}{2gY_sU_s}$

Table D-2

EVANGELISTI CURVES

Figure	Chart	Process		Throttling
		Type	n	
D-3	1	Isothermal	1.0	None
D-4	2	Adiabatic	1.41	"
D-5	3	Isothermal	1.0	Normal
D-6	4	Adiabatic	1.41	"

is the absolute static head. Charts 3 and 4 (Figures D-5 and D-6) indicate the reduction in surge resulting from a symmetric throttling orifice (with same loss factor for flow in either direction) which is proportioned to give a maximum initial drop just equal to the final maximum downsurge as depicted in Figure D-2-(b); this condition has been entitled normal throttling. Throttling losses greater or less than this amount may be said to give overthrottling or underthrottling, respectively.

All four of these charts are based on an assumed rigid water column and are therefore practically limited to values of ρ^* greater than, say, unity. The effects for smaller ρ^* are as indicated in the authors' charts and in Figure D-1 for K (or h_o) equal to zero.

It is evident from these graphs that the greatest effect of elasticity is upon the minimum pressure at the midlength, where the downsurge only approaches one-half the value of the downsurge at the pump if ρ^* is very large; otherwise the decelerating (minimum) design gradeline along the pipeline is distinctly concave.

The writer is disturbed by the fact that the authors' charts seem to be based on the extreme heads in the pipe; if the downsurge curves are based on values of minimum head in the pipe, then they cannot be used to determine the maximum air expansion in the chamber for the overthrottled region of operation. The writer wishes clarification from the authors on these points.

It should be noted that on the Evangelisti charts the surge values given are always for extreme head fluctuations in the air chamber; these will always correspond to the extreme values in the pipeline so long as the system is not overthrottled.

Example

An example may be given of the use of these Evangelisti charts, first to corroborate the results of the authors' example and secondly to provide information for alternative designs.

Let us again assume the plant of the authors' Figure 7. A chamber is desired to limit the maximum head at the pump to 300 feet and the minimum head at the pump to 130 feet.

The significant physical quantities may then be computed as follows:

$$Y_S = 197 + 34 = 231 \text{ feet}$$

Allowable upsurge at the pump is

$$300 - 197 = 103 \text{ feet} = 0.45\ Y_S$$

Allowable downsurge at the pump is

$$197 - 130 = 67 \text{ feet} = 0.29\ Y_S$$

The value of the friction loss has been estimated by the authors to be

$$H_O = 3 \text{ feet} = 0.01\ Y_S$$

Therefore considering only downsurge values, and using $z_{min.} = -0.29$ and $h_O = 0.01$ one finds the following required values of τ from the Evangelisti Charts:

From Chart 1: $\sigma = 0.07$
From Chart 2: $\sigma = 0.04$
From Chart 3: $\sigma = 0.13$
From Chart 4: $\sigma = 0.09$

Since the authors have assumed a process exponent n = 1.2, an average of the first two and last two values of σ represent approximately the required σ values for this condition.

This yields the values

NO THROTTLING: $\sigma = 0.055$
NORMAL THROTTLING: $\sigma = 0.11$

These may be converted directly into air volumes under static conditions through relation (8) of Table D-1, to give

$$U_S = \left(\frac{V_O^2 AL}{2gY_S}\right)\frac{1}{\sigma} = \frac{68.3}{\sigma}$$

which gives for the above throttling conditions

NO THROTTLING: $U_S = 1240$ ft.^{3}
NORMAL THROTTLING: $U_S = 620$ ft.^{3}

To realize normal throttling it would be necessary to introduce an orifice that would create a head loss of

$$H_O + z_{min} = 3 + 67 = 70 \text{ feet}$$

for the rated flow out of the chamber.

The maximum surges corresponding to these chamber volumes may also be calculated directly from the Evangelisti charts: In this case using the right-hand side of the Figures D-3 to D-6, and the parameter values $h_O = 0.01$ and σ as indicated one obtains:

From Chart 1: With $\sigma = 0.07$; $z_{max} = 0.44$
From Chart 2: With $\sigma = 0.04$; $z_{max} = 0.40$
From Chart 3: With $\sigma = 0.13$; $z_{max} = 0.25$
From Chart 4: With $\sigma = 0.09$; $z_{max} = 0.25$

Again averaging the first two and last two pairs to obtain estimates for n = 1.2, this gives

NO THROTTLING: $Z_{max} = 0.42\ Y_S = 97$ feet
NORMAL THROTTLING: $Z_{max} = 0.25\ Y_S = 58$ feet

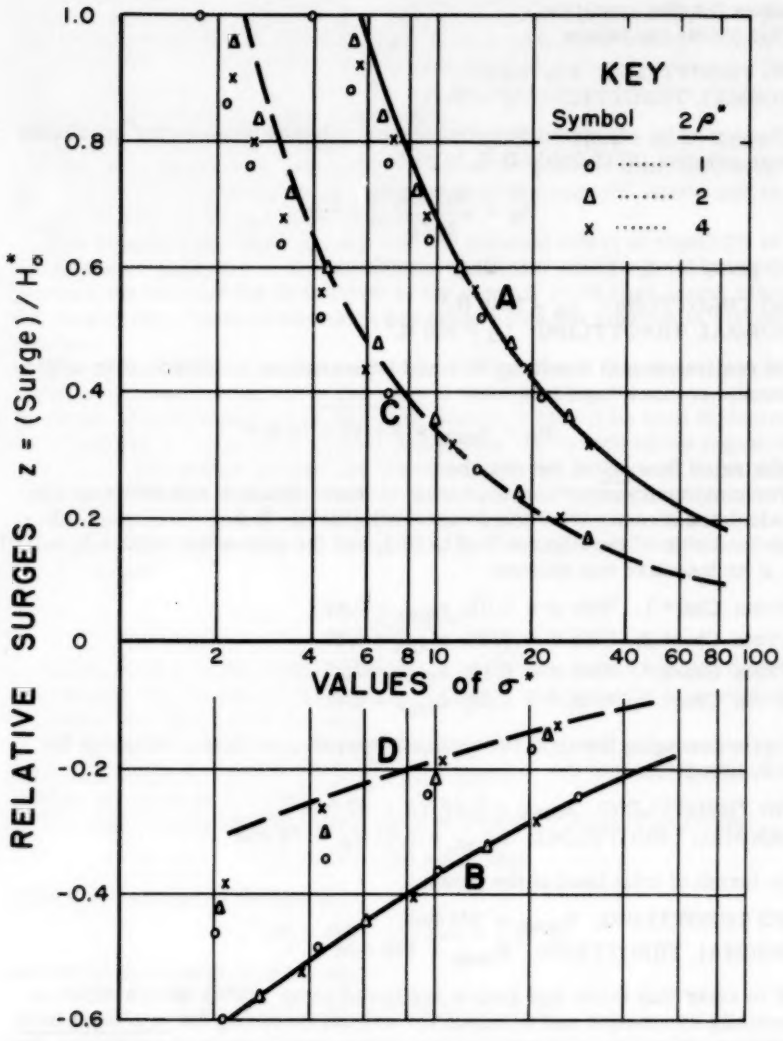
Or in terms of total head at the pump:

NO THROTTLING: $H_{max} = 294$ feet
NORMAL THROTTLING: $H_{max} = 255$ feet

It is clear that these last values are based on an orifice design which is essentially symmetric and designed for normal throttling for minimum head conditions. When designs must be based upon maximum head conditions, the steps must be altered.

CONCLUSION

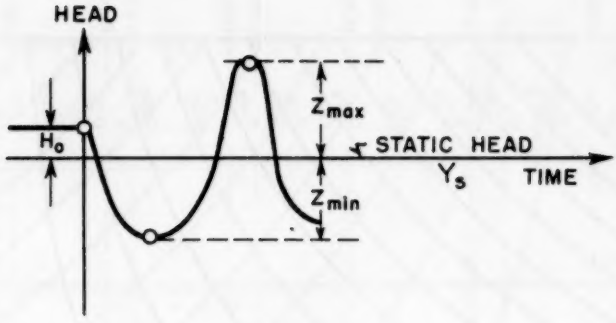
In conclusion, the writer would agree with the authors that their charts and the Evangelisti charts should only be used for preliminary design purposes and that final detailed check computations be made, graphically, numerically or otherwise, using the basic dynamic equations for the system.



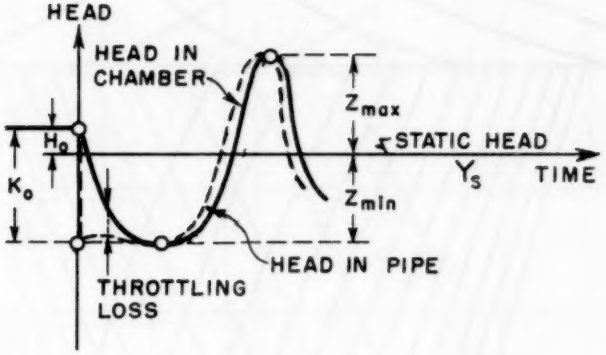
AIR CHAMBER SURGES

K = 0 ; n = 1.2

FIGURE D-1



(a) CHAMBER WITHOUT THROTTLING



(b) CHAMBER WITH NORMAL THROTTLING

AIR CHAMBER PERFORMANCE
(NOTATION OF G. EVANGELISTI)

FIGURE D-2

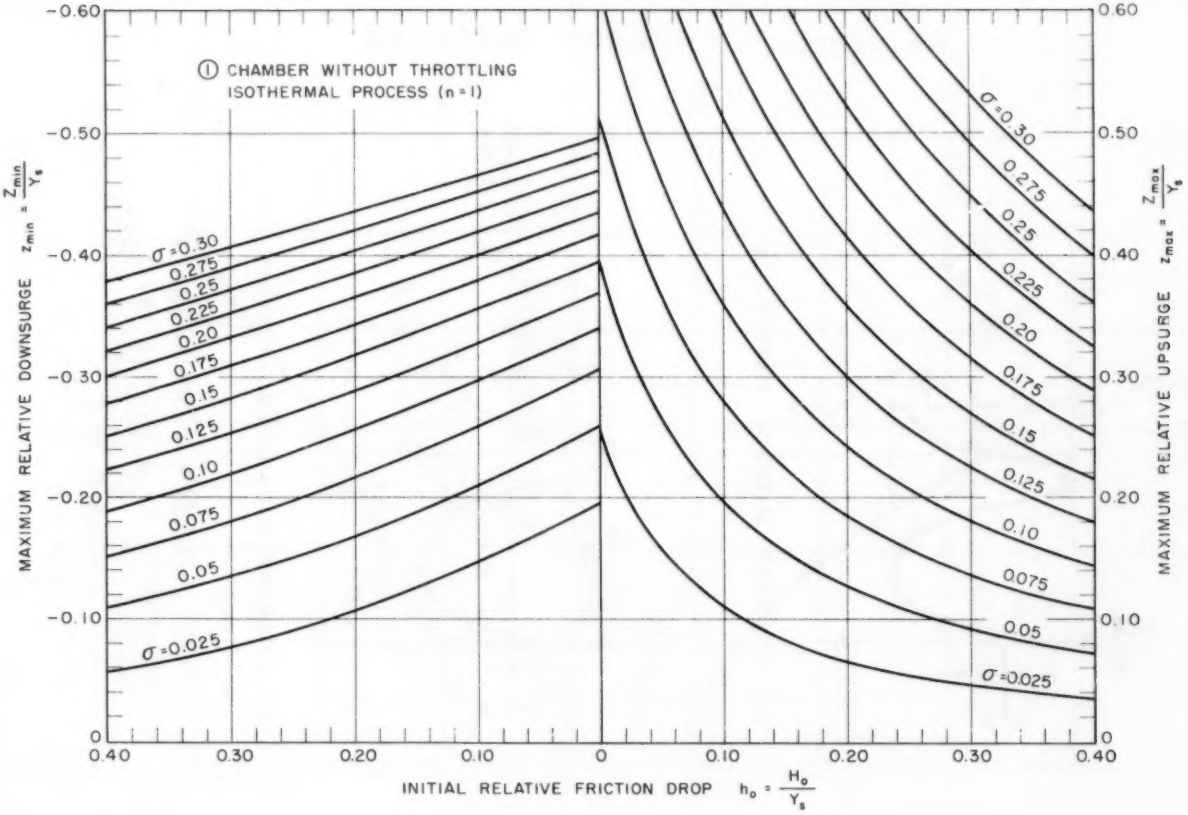

① CHAMBER WITHOUT THROTTLING
ISOTHERMAL PROCESS (n = 1)
MAXIMUM RELATIVE DOWNSURGE $z_{min} = \frac{Z_{min}}{Y_s}$
MAXIMUM RELATIVE UPSURGE $z_{max} = \frac{Z_{max}}{Y_s}$
INITIAL RELATIVE FRICTION DROP $h_o = \frac{H_o}{Y_s}$
σ = 0.30
0.275
0.25
0.225
0.20
0.175
0.15
0.125
0.10
0.075
0.05
σ = 0.025
−0.60
−0.50
−0.40
−0.30
−0.20
−0.10
0
0.60
0.50
0.40
0.30
0.20
0.10
0.40
0.30
0.20
0.10
0
0.10
0.20
0.30
0.40

Fig. D-3

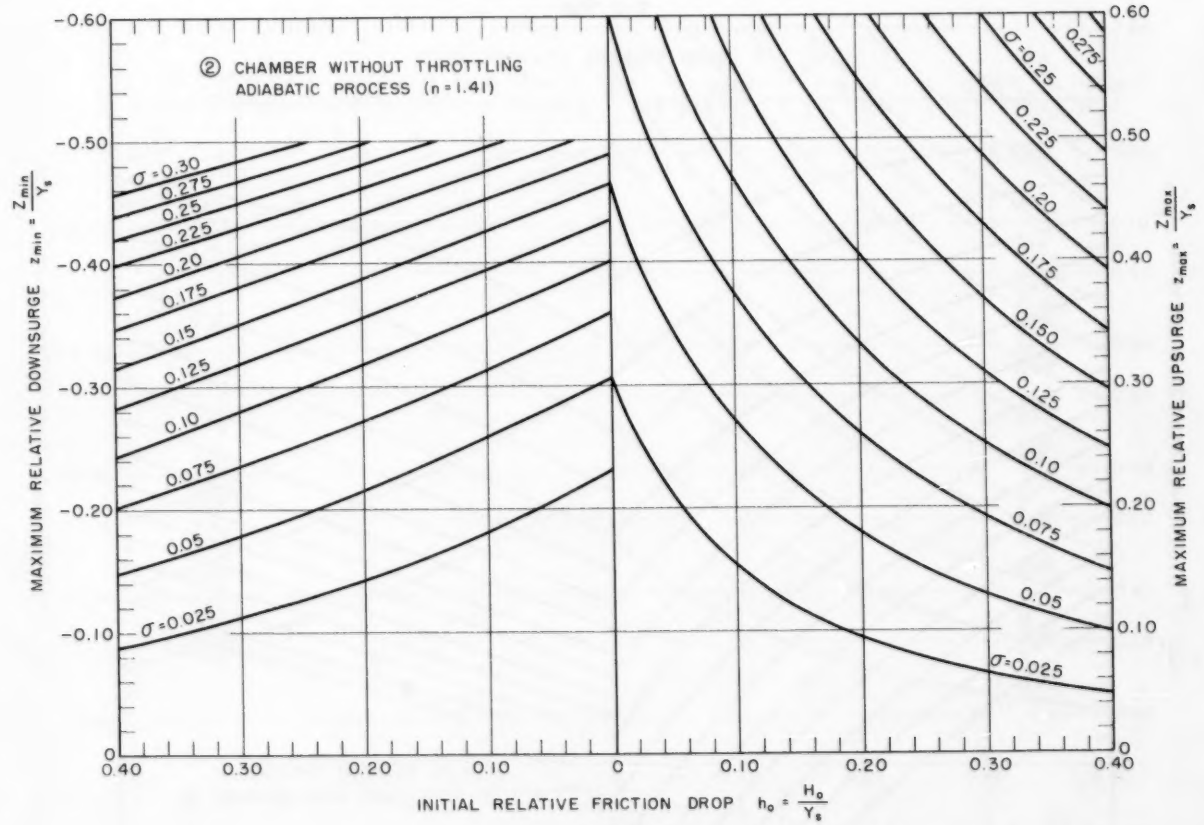

② CHAMBER WITHOUT THROTTLING
ADIABATIC PROCESS (n = 1.41)
MAXIMUM RELATIVE DOWNSURGE $z_{min} = \frac{Z_{min}}{Y_s}$
MAXIMUM RELATIVE UPSURGE $z_{max} = \frac{Z_{max}}{Y_s}$
INITIAL RELATIVE FRICTION DROP $h_o = \frac{H_o}{Y_s}$
σ = 0.30
0.275
0.25
0.225
0.20
0.175
0.15
0.125
0.10
0.075
0.05
σ = 0.025
σ = 0.25
0.275
0.225
0.20
0.175
0.150
0.125
0.10
0.075
0.05
σ = 0.025
-0.60
-0.50
-0.40
-0.30
-0.20
-0.10
0
0.60
0.50
0.40
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Fig. D-4

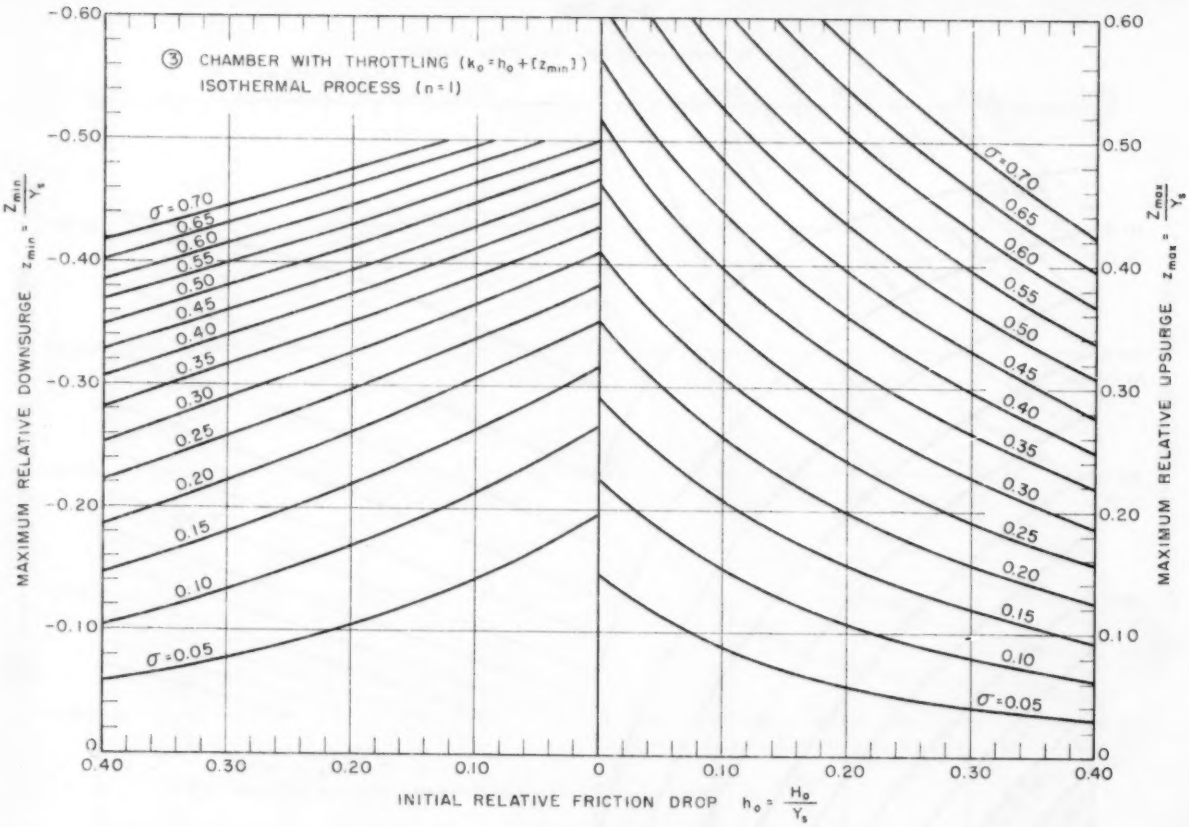
③ CHAMBER WITH THROTTLING ($k_o = h_o + |z_{min}|$)
ISOTHERMAL PROCESS (n=1)
MAXIMUM RELATIVE DOWNSURGE $z_{min} = \frac{Z_{min}}{Y_s}$
MAXIMUM RELATIVE UPSURGE $z_{max} = \frac{Z_{max}}{Y_s}$
INITIAL RELATIVE FRICTION DROP $h_o = \frac{H_o}{Y_s}$
σ=0.70
0.65
0.60
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
σ=0.05
-0.60
-0.50
-0.40
-0.30
-0.20
-0.10
0
0.60
0.50
0.40
0.30
0.20
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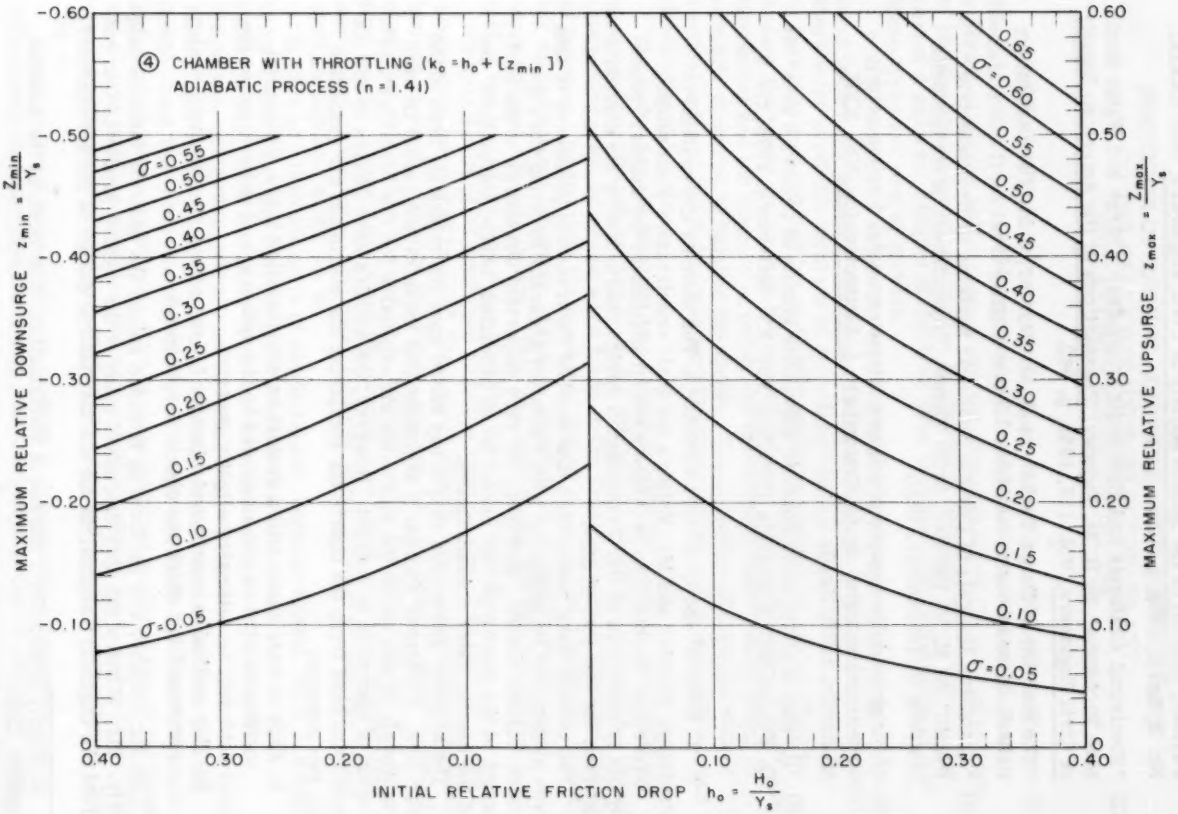
Fig. D-5


④ CHAMBER WITH THROTTLING ($k_o = h_o + [z_{min}]$)
ADIABATIC PROCESS (n = 1.41)
MAXIMUM RELATIVE DOWNSURGE $z_{min} = \frac{Z_{min}}{Y_s}$
MAXIMUM RELATIVE UPSURGE $z_{max} = \frac{Z_{max}}{Y_s}$
INITIAL RELATIVE FRICTION DROP $h_o = \frac{H_o}{Y_s}$
σ=0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
σ=0.05
σ=0.60
0.65
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
σ=0.05
-0.60
-0.50
-0.40
-0.30
-0.20
-0.10
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0.60
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0.40
0.30
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0.20
0.30
0.40

Fig. D-6

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W. E. EVANS[3] and C. C. CRAWFORD[4].— The authors are grateful to Professor Paynter and Mr. Withers for their constructive discussions.

Professor Paynter has provided a useful addition to the authors' paper by his presentation of Dr. Evangelisti's design charts and by his comparisons between the two sets of data.

The authors have recognized that most of the transient pressures in this type system can be predicted with sufficient accuracy by using the rigid-water-column theory. However, the very important downsurge at the mid-length of the discharge line cannot be so predicted; hence, the authors' use of the elastic water column theory.

The authors' downsurge charts are based upon the minimum head in the pipe line. Professor Paynter's argument that these values cannot be used rigorously to determine the maximum air expansion for the "overthrottled" region of operation is correct. However, the authors have chosen to use the minimum head from the charts for computing the maximum air expansion for two reasons:

1. Only in rare cases will a system be overthrottled if the differential orifice is used as recommended by the authors, and in these cases the tank will be estimated slightly oversize.
2. This method is considered adequate for the stated "preliminary design purposes" for which the charts are intended.

No air chamber can perform as intended unless the controls function properly. Mr. Withers has ably discussed many of the control problems that confront the engineer responsible for the mechanical design of the air chamber.

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DISCUSSION OF RECENT ADDITIONS AND IMPROVEMENTS TO THE HALES BAR HYDROELECTRIC PLANT PROCEEDINGS-SEPARATE NO. 294

GEORGE A. JESSOP,[1] M. ASCE.—The building of the Hales Bar Hydro-Electric Plant from its beginning in 1905 until the recent extension covers a period of almost half a century and the development of the art and science of turbine design and construction is strikingly demonstrated. During this period 4 different designs of turbines have been installed and wheels of 3 of these types are now in service.

It is necessary to select the type and size of the power producing equipment before the design of the power house is completed and in 1904 and 1905 there were available turbines of "standard" design only. It was customary at that time to build turbine units with multiple runners on one shaft to increase the speed and reduce the cost of the generator. Usually the shaft was horizontal and the runners in pairs, the discharge of each runner being in opposite direction to balance the thrust. The selection of multiple runner vertical units at Hales Bar was novel to some extent and in line with a trend developing at the time.

The order for the turbines was placed in 1907 but because of unexpected construction difficulties, the plant did not go into operation until 1914. During this period great advances were made in the hydraulic and mechanical design of turbines. By the time the plant was placed in operation, the power producing equipment and the water passages to and from the turbines were outmoded and of obsolete design. This was unfortunate but was beyond the control of the engineers who conceived and built the project.

Originally the plant contained 10 triple runner units, 2 in each of 5 bays. Each unit consisted of two 72" "standard" wheels, one discharging downward and one upward to balance the thrust, and one 63" "standard" wheel mounted above the pair and to be used as a "booster" when the head was low and the flow ample. These wheels had the "inside-type" gate operating mechanism customary at the time. The rated capacity of the 2 lower wheels was about 4950 horsepower at 37' head.

In 1915 and 1916 six units of single runner, vertical wheels, each in a concrete spiral case and with a concrete elbow draft tube, two units in each of three flumes, were installed. The two units are necessarily on the same transverse centerline, but are on different longitudinal center lines, one being further downstream than the other. The upstream wheel is set low and the downstream wheel high so that the spiral cases and draft tubes are separate and distinct water passages. Two of these turbines replaced two of the original triplex turbines and used the old generators operating at 112.5 RPM. The other 4 units were placed in 2 unused bays and had 100 RPM generators. See Fig. 1.

These turbines are essentially modern in design and construction with efficiencies that compare favorably with the best that can be obtained today in new power plants. The rated horsepower, however, is 4150 at 37' head, a

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substantial reduction from the capacity of the original units. The size of these turbines was limited by the width of the bays as originally built.

The weight of the revolving element of each original unit and the downthrust of the "booster" wheel was carried by an oil pressure thrust bearing. Two of these were also used for 2 of the newer single runner spiral case turbines. The experience with these bearings was unsatisfactory and they were replaced with oil bath bearings with the oil surrounding and covering the bearing at atmospheric pressure. Oil bath bearings of satisfactory design and performance first became available about 1913, a year before the turbines were placed in operation, but many years after they were selected and designed. The triplex units 1 to 8, have Gibbs bearings, and the single runner units 9 to 14 have Kingsbury bearings. All these bearings have been giving excellent service for many years.

In 1925 a program was initiated to replace the 8 remaining original turbines with ones having better mechanical construction, higher efficiency and all the capacity that could be installed on the existing discharge passages or draft tubes. To this end, more efficient runners were used and the distributors, or wheel cases, had solid cast iron stay rings, gates with integral stems supported in bronze bushed sockets in the head covers and bottom rings, with levers keyed to the upper end of the stems and with links connecting the levers to the gate operating rings. These distributors, although submerged, had essentially "outside-type", or modern, gate operating mechanisms thus eliminating the original "inside-type" mechanisms. Internal flanges on the draft tubes were cut out in the field so that runners with the largest possible discharge diameter and greatest capacity could be installed. The rated output of the 2 lower wheels was increased to about 5530 horsepower.

All of the original turbines have now been replaced, although it was not until the summer of 1946 that the final change-over was made.

Just upstream from Hales Bar is the Chickamauga plant with ample storage and with a discharge capacity of 40000 C.F.S. As pointed out by the author, it is advantageous to have the discharge at Hales Bar at least equal to that at Chickamauga. To accomplish this, 2 adjustable-blade propeller turbines were installed in an extension of the old plant. Each of these has a maximum capacity of 11000 C.F.S.

To match the Chickamauga discharge, presumably the adjustable-blade turbines are opened to use about 22000 C.F.S. The 6 single runner spiral case turbines discharge about 6900 C.F.S. and there remains about 11100 C.F.S. for the rebuilt triplex units. This is approximately 84 per cent of the maximum capacity of the 8 pairs of wheels which are the principal elements of the triplex turbines.

It appears, then, that it is frequently necessary to have the 6 single runner units, and all of the triplex units in operation with the new adjustable-blade units.

The author is to be congratulated upon the excellent way in which he has so thoroughly covered his subject and he has merited the thanks of the hydroelectric industry.

ADOLF A. MEYER,[2] M. ASCE.—The writer is most gratified by the interest shown in this paper, as evidenced by the valuable supplemental data presented.

Mr. Blee contributes pertinent observations on the development of foundation investigation and treatment since the original Hales Bar installation. He

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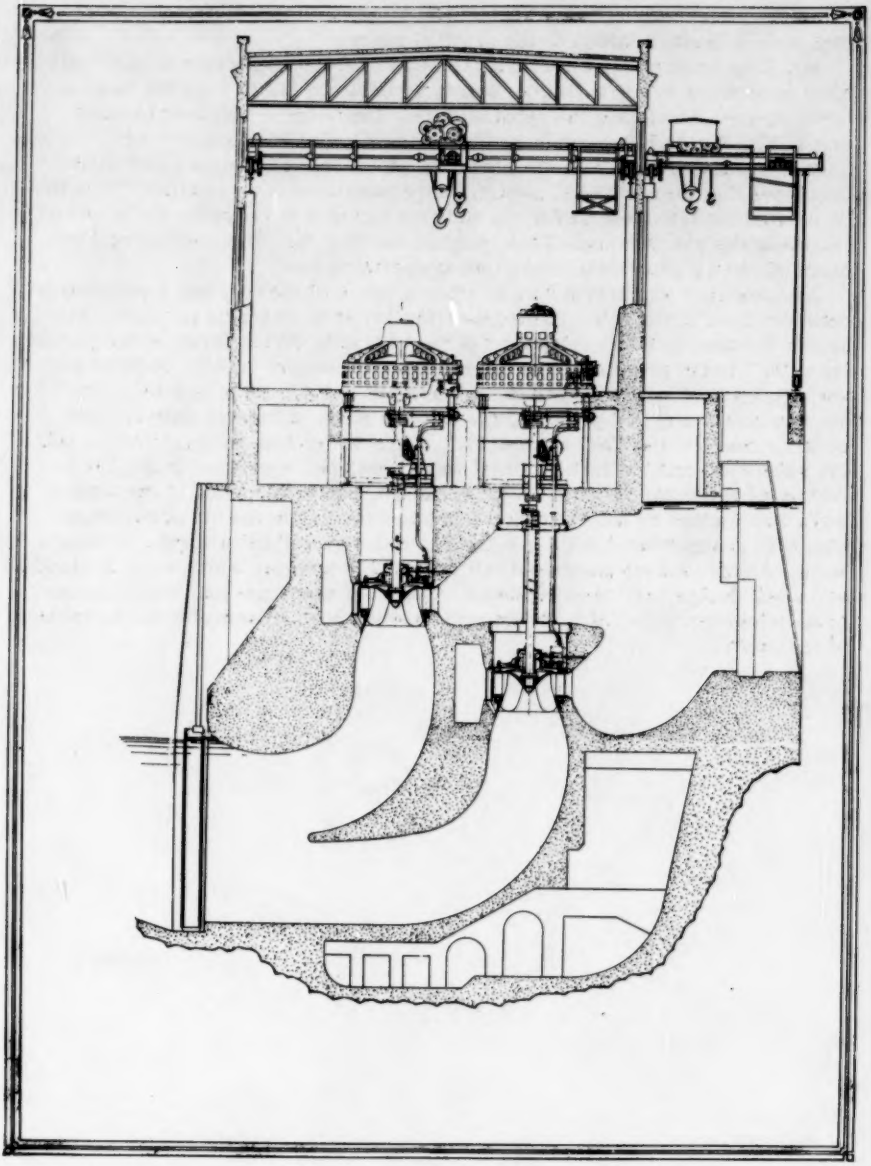

Fig. 1

also adds interesting information on special construction features. These data form a useful addition to the original paper.

Mr. Rich emphasizes the characteristics of the substructure layout: efficient waterways within a simple, sturdy structure. These were the main objectives when developing the substructure. The writer is pleased to know that in Mr. Rich's judgment these objectives were well accomplished.

It is gratifying that from the cost data submitted in the paper Mr. Rich concludes that the additional capacity is economically well justified. It is to be pointed out that the cost for the addition includes the expense for modernization of the old units and of the common service facilities, which together accomplished a substantial reduction in operating cost.

Mr. Jessop's discussion fills in a blank space of the original paper, which confined itself strictly to civil engineering features as far as possible. Mr. Jessop focuses on the development of the hydraulic turbines during the period from 1907 to the present and on its influence on station layout. Unquestionably, Hales Bar, with turbines designed in 1907, 1915, 1925, and 1951, has become something of a prize exhibit on the history of turbine design. Two of these designs, the 1907 and the 1951, were integrated designs; that is, both the waterways and the turbine itself were developed together. In the 1915 design, of which Mr. Jessop added an exhibit, the development of the waterways was limited by the existence of the turbine bay in the old powerhouse. The 1925 design was strictly mechanical redesign without changing of waterways. As Mr. Jessop points out, all of the 1907 turbines which were of standard stock design have been replaced. The three other designs are all in operation together now. Mr. Jessop's discussion adds measurably to the value of the paper.